# SYSTEM FOR ADJUSTING RESONANCE FREQUENCIES IN A LINEAR COMPRESSOR

## Field of the Invention

The present invention refers to a system for controlling and adjusting the resonance frequencies in a linear compressor of the type used in small refrigeration appliances, such as refrigerators, freezers, water fountains, etc.

## Background of the Invention

Linear compressors present a mechanical resonance frequency that is defined by the spring constant and by the masses of the movable components of said compressor when the latter is working with no gas load to be refrigerated (unloaded), i.e., with no gas being pumped. The mechanical resonance frequency of the mass-spring system of the compressor is a function of the mass-spring project thereof and defines the natural mechanical resonance frequency of the compressor.

The mechanical resonance frequency during the operation of the compressor, with the latter pumping gas under a determined pressure ratio, said pressure ratio being defined as the discharge pressure divided by the suction pressure, is influenced by the gas-spring effect resulting from the compression of the refrigerant fluid in the compression chamber of the compressor, which effect is higher or lower, depending on the pressure/dead volume ratio.

The mass-spring system of the compressor is designed to present a mechanical resonance frequency substantially close to the electrical supply frequency of the power system, i.e., about 50Hz or 60Hz, depending on the location. The attainment of this objective may be called operational tuning.

In the tuning condition between the mechanical

resonance frequency of the compressor and the electrical supply frequency of the power system, the energy that must be supplied to the compressor is basically the sum of the energy consumed for gas compression and the energy consumed with friction between the movable parts in operation.

The known compressor designs usually present a mass-spring system with a natural mechanical resonance frequency, without considering the loads below the electrical supply frequency of the power system.

In order to be efficient during operation in a refrigeration system, the linear compressor must have its mechanical resonance frequency equal or at least substantially equal to the electrical frequency supplied to the motor of the compressor, since in this condition there is a balance between the accumulation and liberation of energy, establishing a tuned operational condition. When the compressor is working out of tune, it needs to receive more energy to keep it running and also to generate compression work.

The gas being pumped during the operation of the compressor in a refrigeration system acts as an additional spring in the mass-spring system of the compressor, modifying its mechanical resonance frequency, conducting the latter to values that can diverge upwardly or downwardly from the value of the electrical supply frequency of the power system. This additional spring, or gas-spring, presents an average constant which is a function of the pressure/dead volume ratio. When gas is compressed outwardly from the cylinder, part of the compression energy returns to the mechanical system, delivering work back to the mechanical system, resulting in a spring effect. In pressure ratio increase conditions, the gas-spring effect is intensified, increasing the mechanical

resonance frequency. In dead volume increase conditions, upon reducing the capacity of the compressor, the gas-spring effect is also intensified, increasing the mechanical resonance frequency.

In the refrigeration system, the pressures depend on the thermal load existing inside the refrigeration appliance, i.e., the source of thermal load existing in the interior thereof and which is generating heat, which the system has to remove; and on the temperature of the environment where the refrigerator is located, since, if the temperature of this environment is high, the temperature of the condenser must be higher than the temperature of the environment, since the condenser must work transferring heat to the environment that is external to that being refrigerated by the refrigeration appliance.

In warmer days, as well as in conditions in which more load is to be refrigerated, the gas compression pressures rise and the compressor is forced to work more so that the refrigeration system can remove heat from the medium under refrigeration. The pressure variations in the refrigeration system vary the capacity of the compressor and modify the tuning condition between its mechanical resonance frequency and the electrical supply frequency of the power system.

In determined situations, the loss of balance between the electrical and mechanical frequencies in the compressor results, for the latter, in a higher need of energy to maintain the pumping of gas.

In a situation in which the refrigeration appliance is conducted from a turn-off condition to a turn-on condition, the refrigeration system of this appliance is submitted to high pressure peaks, with maximum pumping capacity, which increases the mechanical

resonance frequency of the compressor, causing unbalance between said mechanical resonance frequency and the electrical supply frequency of the power system. In such conditions, the motor of the compressor has to increase its operational force to keep the mechanism in the same frequency as that of the power system. Since the motor yield is a function of its effort to operate, there is loss of motor efficiency whenever the mechanical resonance frequency is not equalized with the electrical supply frequency of the power system.

There are known from the prior art solutions to adjust the electrical supply frequency with the mechanical resonance frequency of the compressor under operation. In one of said solutions, the frequency adjustment is obtained by electronic balance, rectifying the frequency of the power system and then altering the latter according to the changes that occur in the mechanical system.

In a particular solution within this concept, the electronic balance is achieved by varying the velocity of the compressor motor (Brazilian patent document PI9601535-7). However, this solution is expensive and presents energy loss.

There are not known prior art solutions for adjusting the mechanical resonance frequency, which allow, during operational conditions, correcting said frequency and making it substantially equal to the electrical supply frequency of the power system.

## Objects of the Invention

Thus, it is an object of the present invention to provide

a system for adjusting the resonance frequencies in a linear compressor, which controls and conducts the mechanical resonance frequency of the compressor, at

least in determined operational conditions, to values that are substantially close to the electrical supply frequency of the compressor, without presenting the high costs and high energy losses of the electronic control systems.

It is a further object of the invention to provide a system such as mentioned above, which maintains the dead volume of the compressor at a minimum value during the operation thereof, maintaining an adequate volumetric yield, with minimum energy loss.

## Disclosure of the Invention

These and other objects are achieved through a system for adjusting the mechanical resonant frequencies in a linear compressor comprising, in the interior of a shell: a linear motor supplied by an AC electrical current presenting a predetermined electrical supply frequency; a cylinder, within which is defined a compression chamber closed by a valve plate; and a resonant assembly defined by a piston reciprocating inside the cylinder in consecutive suction and compression strokes, and an actuating means operatively coupling the piston to the linear motor.

The system for adjusting the mechanical resonance frequency of the present invention comprises: a detecting means to detect a load imposed to the linear motor of the compressor, in an operational condition of the latter related to the gas pressure in the discharge thereof; and a frequency adjusting means operatively associated with the detecting means and with the resonant assembly, in order to define, as a function of the operational condition detected for the gas in the discharge of the compressor, a frequency adjustment by varying at least one of the values related to the mass of the resonant assembly and to the average stroke of the piston, to a value of the

mechanical resonance frequency of the resonant assembly corresponding to the electrical supply frequency, maintaining unaltered the minimum distance between the piston and the valve plate at the end of each compression stroke.

Brief Description of the Drawings

The invention will be described below, with reference to the attached drawing, in which:

Figure 1 is a simplified schematic longitudinal vertical sectional view of a linear compressor to which the present invention can be applied;

Figure 1a is a schematic view, similar to that of figure 1, but illustrating another embodiment of the present solution;

Figure 2 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a first embodiment of the invention, according to which the piston dead point, opposite to the valve plate, is varied by a hydraulic or pneumatic impeller;

Figure 3 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a second embodiment of the invention, in which the piston dead point, opposite to the valve plate, is varied by a hydraulic or pneumatic impeller;

Figure 4 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a third embodiment of the invention, in which the piston dead point, opposite to the valve plate, is varied by a mechanical impeller in the form of a cam with linear displacement;

Figure 4a is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating another embodiment for the cam with linear displacement of the third embodiment of the invention:

Figure 5 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a fourth embodiment of the invention, in which the piston dead point, opposite to the valve plate, is varied by a mechanical impeller in the form of a rotary cam;

Figure 6 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a fifth embodiment of the invention, in which the piston dead point, opposite to the valve plate, is varied by a mechanical impeller in the form of a screw shaped mechanical stop means;

Figure 7 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a sixth embodiment of the invention, in which the piston dead point, opposite to the valve plate, is varied by a pneumatic impeller with a particular construction; and

Figure 8 is a partial schematic longitudinal sectional view of the compressor of figure 1, illustrating a seventh embodiment of the invention, in which the variation of the adjustment of the resonance frequencies of the present invention is obtained by varying the mass in the interior of the piston.

## Description of the Illustrated Embodiments

The present invention will be described in relation to a reciprocating compressor driven by a linear motor of the type used in refrigeration systems and comprising, within a hermetic shell (not illustrated), a motor-compressor assembly including a cylinder 1, which is closed, at one of the ends thereof, by a valve plate 2, and inside which is provided a piston 10 reciprocating in consecutive suction and compression strokes.

In conventional constructions, an internal lower

portion of the shell defines a reservoir for the lubricant oil of the compressor.

In the valve plate 2 there are defined a suction orifice 3 and a discharge orifice 4 of the compressor, which are respectively and selectively closed by a suction valve 5 and by a discharge valve 6, in order to allow the selective fluid communication between a compression chamber CC, which is defined within the cylinder 1 between a top portion of the piston 10 and the valve plate 2, and respective internal portions of a cylinder head 7 which are maintained in fluid communication with the low pressure and high pressure sides of the refrigeration system including the compressor.

As illustrated in the appended drawings, the compressor further comprises a linear motor 20, mounted around the cylinder 1 and the piston 10 and including a stack of internal laminations 21 with a magnet 22 inserted therein and which is axially impelled upon energization of the linear motor 20, and a stack of external laminations 23.

In the illustrated construction, the compressor further includes a conventional spring means 8, coupling a resonant assembly to a non-resonant assembly C of the compressor and which is elastically axially deformable in the displacement direction of the piston 10, and an actuating means 9 that carries the magnet 22, said actuating means 9 operatively coupling the piston 10 to the linear motor and defining, with said piston 10 and spring means 8, a resonant assembly of the compressor.

The linear motor 20 is supplied by an electrical current presenting an electrical supply frequency that is previously determined, for example around 50 or 60Hz, which generally corresponds to the electrical

supply frequency of the power system.

According to the present invention, the adjustment between the mechanical resonance frequency of the compressor and the electrical supply frequency thereof is achieved through a system for adjusting the frequencies which comprises, generically, a detecting means D for sensing a load imposed to the linear motor 20 of the compressor in an operational condition thereof and which is related to the pressure of the gas at its discharge; a frequency adjusting means, which is operatively associated with the detecting means D and with the resonant assembly, so as to define, as a function of the operational condition detected for the gas in the discharge of the compressor, for example, at least one of the conditions of: pressure and temperature of the gas compressed in the discharge of the compressor, and operational electrical current of the linear motor 20, a frequency adjustment, by varying at least one of the values related to the mass of the resonant assembly and to the average stroke of the piston 10, to a value of the mechanical resonance frequency of the resonant assembly corresponding to the electrical supply frequency, maintaining unaltered the minimum distance between the piston 10 and the valve plate 2 at the end of each compression stroke.

In one embodiment to be described below, the adjusting system of the present invention comprises a control unit 30, operatively connected with both the detecting means D and the adjusting means, in order to receive, from the former, information about one of the operational conditions of: pressure and temperature of the gas in the discharge of the compressor, and operational electrical current of the linear motor 20, in order to instruct the adjusting means to provide

one of the operations related to varying the average stroke of the piston 10 and varying the mass of the resonant assembly.

In a refrigeration system, the pressures depend on the thermal load found inside the refrigeration appliance generating heat, which the refrigeration system has to remove and which defines the temperature of the medium; the higher the temperature of the medium, the higher must be the temperature of the condenser so as to transfer heat to said medium. The pressures in the refrigeration system change continuously and to compensate such changes, it is necessary to vary the capacity of the compressor.

In a situation in which the refrigeration appliance, for example a refrigerator, is conducted from a turn-off condition to a turn-on condition, the refrigeration system is submitted to a high pressure peak, increasing the mechanical resonance frequency.

On the other hand, variations in the load imposed to the linear motor 20 provoke a variation in the phase/intensity of the current, with the dynamics of the mechanism being defined by one of the following parameters: displacement, velocity, or acceleration.

The adjustment of the mechanical resonance frequency to the electrical supply frequency can be achieved by:

- varying the discharge pressure/suction pressure ratio, considering that rises of the pressure ratio increases the mechanical resonance frequency;
- varying the stroke of the piston 10, considering that increases of the stroke reduce the mechanical resonance frequency;
- varying the dead volume, considering that the rise of the dead volume increases the mechanical resonance frequency: and
- varying the mass of the piston 10, considering that

the increase of the mass of the piston 10 reduces the mechanical resonance frequency.

The present invention provides a system for adjusting the resonances, which uses at least one of the values related to the average stroke of the piston 10 and to the mass of the resonant assembly of the compressor, tuning the compressor for working in a high pressure ratio condition, in order to overcome the critical conditions required by the refrigeration system, with a minimum dead volume.

According to the constructive forms of the invention illustrated in figures 2-7, the desired adjustment for the mechanical resonance frequency of the resonant assembly is effected by an adjusting means that varies the operational average stroke of the piston 10, obtained by modifying the dead point of the piston 10 at the end of the suction stroke.

In the embodiments illustrated in figures 2-6, the modification of the dead point of the piston 10 at the end of the suction stroke is effected by an adjusting means in the form of an impeller I, which can be defined, for example, by an hydraulic actuator, a pneumatic actuator, and a mechanical actuator which is operatively coupled to the resonant assembly and to the control unit 30, so as to be driven by the latter between an inoperative condition, in which it does not produce any alteration in the stroke of the piston 10, and an operative condition, in which it modifies the stroke of the piston 10 for adjusting the mechanical resonance frequency of the resonant assembly to the electrical supply frequency.

In the embodiment of figure 2, in which the impeller I is a hydraulic actuator 40, the latter has a cylinder C1 defined in a portion of the non-resonant assembly C of the compressor, and a plunger operatively coupled

to the spring means 8, the hydraulic actuator 40 being maintained in direct fluid communication with a reservoir of equalizing fluid provided in the interior or exterior of the compressor shell, through an adequate duct D1. In one embodiment of the present solution, the equalizing fluid is the lubricant oil of the compressor.

In the embodiment in which the impeller I is a pneumatic actuator 50, the latter can be also constructed as described in relation to the hydraulic actuator 40 illustrated in figure 2, by only substituting the non-compressible equalizing fluid for a compressible fluid, such as gas. In one constructive solution, the gas that activates the pneumatic actuator 50 is the refrigerant gas that existing inside the shell.

In one embodiment of the present invention illustrated in figure 3, the pneumatic actuator 50 is in the form of a bellows operatively coupled to the non-resonant assembly C and to the spring means 8.

As it can be noted, the higher or lower internal pressurization of the hydraulic or pneumatic actuator allows a predetermined axial displacement of the respective piston 41 or 51 to be obtained, in order to cause a corresponding modification of the suction dead point of the piston 10 at the end of the suction stroke, which is necessary to produce a variation in the mechanical frequency of the resonant assemble that is capable of compensating the variations of said frequency as a function of the modifications of the operational conditions of the refrigeration system. This variation of the stroke of the piston 10 is effected so as to keep unaltered the dead point of the piston at the end of the compression stroke, that is, the dead volume in the compression chamber CC.

According to the illustrations in figures 4-6, the impeller I is a mechanical actuator, operatively coupled to the non-resonant assembly C and to the spring means 8 and which is operated by a driving means M, in the form of a motor or a hydraulic or pneumatic actuator, which moves said mechanical actuator to different operational positions.

In one embodiment of the present invention, the driving means M is an electric motor operatively connected to the control unit 30, so as to receive, from the latter, instructions to vary the average stroke of the piston 10.

In the embodiment illustrated in figure 4, a mechanical actuator 60 is in the form of a cam of linear displacement 61, which is for example provided with steps 62 which are dimensioned to define different positions for the dead point of the piston 10 at the end of the suction stroke. In the illustrated construction, the cam of linear displacement 61 has two steps 62 defining two different positioning levels for the dead point of the piston 10 at the end of its suction stroke, said steps 62 being joined to each other through a ramp surface 63.

In this embodiment, the cam of linear displacement 61 acts against a slide 64, axially movable and which defines a cam follower that is seated on the spring means 8 of the resonant assembly C. The slide 64 has its axial displacement effected in the interior of the guide means 65 incorporated to the non-resonant assembly C.

In this illustrated embodiment, the slide carries a contact portion 64a, such as a portion with a convex surface that is incorporated to the surface of the slide 64 confronting with the cam of linear

displacement 61, said convex surface being a spherical calotte, as illustrated.

Figure 4a illustrates a different construction for the present solution, in which a mechanical actuator 60 is in the form of a cam of linear displacement 61´ presenting a ramp surface 63´ which is slidingly seated against a confronting inclined surface 66 of an axially moving slide 64´ which defines a cam follower, such as described in relation to the slide 64. As described in relation to the construction illustrated in figure 4, in this construction the slide 64´ is seated on the spring means 8 of the resonant assembly.

In another construction, as illustrated in figure 5, a mechanical actuator 70 is in the form of a rotary cam 71 presenting a continuous ramp surface 71a, which is dimensioned so as to define, continuously, different positions for the dead point of the piston 10 at the end of the suction stroke, said rotary cam 71 being mounted to an adjacent portion of the non-resonant assembly C and acting against a slide 72 defining a cam follower seated on the spring means 8 of the resonant assembly, said slide 72 being also provided with a continuous ramp surface 73, against which the continuous ramp surface 71a of the rotary cam 71 is slidingly seated.

In the construction illustrated in figure 6, the mechanical actuator is in the form of a mechanical stop means 80, threaded to the non-resonant assembly C and operatively coupled to the resonant assembly, in order to alter the dead point of the piston 10 at the end of the suction stroke, upon being rotated around its longitudinal axis by any driving means M.

In these constructive solutions illustrated in figures 2-6, as well as in figure 8 to be discussed below, the modifications of the dead point of the piston 10 at

the end of the suction stroke are commanded by a control unit 30, as a function of the information the latter receives from the detecting means D.

According the constructive form of the present invention illustrated in figure 7, the dead point of the piston 10 at the end of the suction stroke is automatically modified by the gas pressure variations in the discharge of the compressor.

In this construction, the impeller I is a pneumatic actuator 90, constructed for example as described in relation to the hydraulic actuator illustrated in figure 2 and which presents a cylinder 91, incorporated to the non-resonant assembly C, and a plunger 92 axially displaceable within the cylinder 91 and operating as a movable stop means onto which the spring means 8 of the resonant assembly will be seated.

In this construction, the displacement of the plunger 92 is obtained by the higher or lower pressurization of the cylinder 91, by means of the refrigerant gas that is used in the refrigeration system.

In the construction illustrated in figure 7, the closed end of the cylinder 91 is provided with at least one opening 93 maintained in fluid communication with the interior of the body of a control valve 100 lodging a sealing means 110 which is selectively displaced between a closed position, a pressurization position, and depressurization position, in order to, respectively, block the opening 93 of the cylinder 91 upon discharge of the compressor and communicate the interior of the cylinder 91 with the interior of the compressor shell.

The control valve 100 presents at least two passages 101, one of which being opened to the interior of the compressor shell and the other of said passages 101

being defined opened to a respective opening 93 of the closed end of the cylinder 91, so as to allow, selectively, the fluid communication between the interior of the cylinder 91 and the interior of the compressor shell, as a function of the displacement of the sealing means 110 in the interior of control valve 100.

The closing of the opening 93 by the sealing means 110 allows the plunger 92 to be maintained in a stable position, defining a determined stroke for the piston 10. Upon occurring a rise in the discharge pressure, the sealing means is automatically displaced to the depressurization position, in order to communicate the interior of the cylinder 91 with the compressor shell, promoting depressurization thereof, sufficient to correspondingly reduce the mechanical resonance frequency, by increasing the average course of the piston 10. The plunger 92 is axially displaced towards the closed end of the cylinder 91.

On the other hand, upon occurring a reduction in the discharge pressure of the compressor, the sealing means 110 is automatically displaced to the pressurization position, communicating the interior of the cylinder 91, by aligning an opening 93 and a passage of the control valve 100, with the discharge of the compressor, promoting a degree of pressurization of the cylinder 91 sufficient to displace the plunger 92 toward the open end of the cylinder 91, thereby reducing the average stroke of the piston 10 and consequently increasing the mechanical resonance frequency to compensate for the reduction in the latter caused by the discharge pressure drop.

The displacement of the sealing means 110 between its different operational positions is achieved, in a

first direction, by the discharge pressure itself, which is acting on the sealing means 110 in a direction opposite to that of the force produced by the discharge gas pressure.

In the illustrated example, the sealing means 110 takes the form of a slide provided with an internal passage 111 and which is linearly displaced, in one and in the other direction, by the discharge gas pressure and by a return elastic means 20, in order to align or disalign said passage 101 of the sealing means 110 in relation to each passage 101 provided in the construction of the illustrated control valve 100.

In the constructions presenting modifications in the operational average stroke of the piston 10, obtained by varying the dead point of the piston 10 at the end of the suction stroke, the present solution implies that upon being detected, for example a temperature (or pressure) rise of the gas compressed by the compressor, the impeller means I, automatically or by instruction of the control unit 30, acts on the resonant assembly, so as to reduce the average stroke of the piston 10 by a sufficient value to cause a corresponding rise in the mechanical resonance frequency of the resonant assembly of the compressor, until the mechanical resonance frequency is adjusted to the electrical supply frequency of the compressor.

In the case of temperature (or pressure) reduction of the gas compressed by the compressor, the impelling means acts on the resonant assembly, in order to increase the average stroke of the piston 10, reducing, consequently, the mechanical resonance frequency of the compressor. These variations of the average stroke of the piston 10 are effected so as to maintain unaltered the dead point of the piston 10 at the end of the compression stroke, i.e., the dead

volume in the compression chamber CC.

In the embodiment illustrated in figure 8, the frequency adjustment of the compressor is achieved by an adjusting means, which varies the mass of the resonant assembly, for example, by varying the mass of at least one of the parts defined by the piston 10 and the actuating means 9. In this solution, the modifications are commanded by the control unit 30, as a function of the information received from the detecting means.

According to the present invention, each part of the resonant assembly to have its mass modified comprises an internal chamber 11 containing an equalizing fluid and which is maintained in fluid communication with an equalizing fluid reservoir defined in the interior or in the exterior of the compressor shell, the variation of the mass of the resonant assembly being obtained by altering the mass of the fluid inside the internal chamber.

In the illustrated construction, the variation of the mass of the resonant assembly is obtained by varying the mass in an internal chamber 11, for example presenting a constant volume and being defined in the piston 10, said internal chamber 11 being maintained in fluid communication with an equalizing fluid impelling means 130 provided inside the compressor shell in fluid communication with the equalizing fluid reservoir, so as to, selectively, pump said equalizing fluid into and out from said internal chamber 11, by instruction of the control unit 30.

In one embodiment of the present invention, the equalizing fluid is defined by the lubricant oil of the compressor provided in the oil reservoir defined at the bottom of the compressor shell.

According to this embodiment, when the detecting means

informs the control unit 30 that there occurred a value variation in the parameter being analyzed, for example in the temperature of the gas compressed in the discharge of the compressor, the control unit 30 instructs the equalizing fluid impelling means 130 to displace a respective regulator-actuator 131, with the purpose of adding to or removing from the internal chamber 11 a determined amount of the equalizing fluid which is sufficient to permit a variation of mass which compensates for the variation of the resonance frequency caused by the variation of the gas discharge pressure, said condition being maintained by instruction of the control unit 30 until the detecting means informs the value of the considered parameter has reached a value that corresponds to the normal operation of the compressor.

The present solution implies that, upon detecting for example a temperature (or pressure) rise of the gas compressed by the compressor, the control unit 30 instructs the equalizing fluid impelling means 130 to remove a determined amount of equalizing fluid from the internal chamber 11 of the piston 10, sufficient to allow reducing the mass of the latter by a value which results in a determined increase of the mechanical resonance frequency of the resonant assembly of the compressor. The control unit 30 instructs the equalizing fluid impelling means 130 to insert a determined quantity of equalizing fluid into the internal chamber 11, increasing the mass of the resonant assembly, upon receiving information from the detecting means that the temperature (or pressure) of the compressed gas has reached a determined value which corresponds to a modification of the mechanical resonance frequency of the resonant assembly, due to an increase thereof. In this case, the instruction to

increase the mass of the resonant assembly will determine, as a consequence, a reduction of the mechanical resonance frequency. These variations in the mass of the resonant assembly, particularly illustrated in relation to the piston 10, are effected by maintaining unaltered the dead point of the piston 10 at the end of the compression stroke, i.e., the dead volume in the compression chamber CC.

While some ways of carrying out the present invention have been described and illustrated, it should be understood that other embodiments are possible within the inventive concept defined in the appended claims.